2000-01-0108

Performance Characterization and Model Verification of a Loop Heat Pipe

Michael Pauken and José I. Rodriguez
Jet Propulsion Laboratory, California Institute of Technology

ABSTRACT

A simple Loop Heat Pipe (LHP) with a single evaporator and condenser was tested and modeled with two different working fluids: ammonia and propylene. While ammonia exhibits many desirable heat transfer characteristics, its freezing point is too high to prevent freezing in the condenser lines during a safing mode on a satellite platform. Consequently, propylene makes a good compromise since it has a lower freezing point and relatively good heat transfer properties.

The performance of the LHP with ammonia was characterized by a series of tests with heat loads of 20 to 800 watts placed on the evaporator. With the LHP filled with propylene, it was tested with heat loads of 20 to 200 watts to the evaporator. The sink temperatures on the condenser ranged from -10°C to 20°C. The constant conductance performance of the LHP was 170 W/K with ammonia and 44 W/K with propylene.

Steady state performance data of the LHP was used to validate a nodal network model of the device. The evaporator temperature as a function of heat load was compared between the collected data and the model. The average difference between the observed and the predicted evaporator temperatures were 0.85°C with ammonia and 1°C with propylene.

INTRODUCTION

Managing thermal loads on spacecraft requires extremely reliable systems capable of operating under a wide range of conditions. These requirements can be met with Capillary Pumped Loops (CPL)s. However, CPLs require a lengthy preconditioning period to initiate operation and are subject to depriming if vapor builds up in the wick core. Loop Heat Pipes (LHP)s are a variation of the CPL theme that do not need preconditioning for startup and they are able to operate with vapor in the core. These advantages promoted the selection of LHPs as the primary heat transfer device in the Tropospheric Emission Spectrometer instrument which will be flown on NASA's Earth Observing System Chemistry spacecraft platform. Consequently, a test program on LHPs was initiated at JPL to determine the performance characteristics of a LHP demonstration prototype. The specific objectives were to identify: (1) the variable and constant conductance regions, (2) startup problems and (3) test a nodal network model's capabilities for describing actual test data in support of designing LHPs for the TES instrument. This paper describes some early results and observations of this test program.

The experiments reported here are with no mass attached to the evaporator. LHPs typically are easier to start without additional mass on the evaporator since the heat input is quickly supplied to the evaporator. This supplies a high heat flux through the evaporator wall to initiate boiling. The results of experiments with a 21 kg mass attached to the evaporator are discussed in a companion paper [1]. Details on the thermal design of the TES instrument and application to LHPs are available in Ref. [2]. It is outside the scope of this paper to review the basic operating principles of CPLs and LHPs. There are good reviews on these topics in Refs. 3-6. Recent work describing experimental and modeling efforts for LHPs include Refs. 7-9.

EXPERIMENTAL APPARATUS AND PROCEDURES

Figure 1 shows a schematic of the Loop Heat Pipe used in the test program. The LHP was instrumented with 40 type K thermocouples, with one used for measuring the ambient air temperature as shown in the figure. The aluminum evaporator saddle was 15 cm long and had two Calrod heaters imbedded within the saddle.

The stainless steel compensation chamber had thin Kapton film heaters attached to the outer surface. These low power heaters could be used to shut off the LHP during a safing event on an orbiting spacecraft. However, they were used in these experiments to

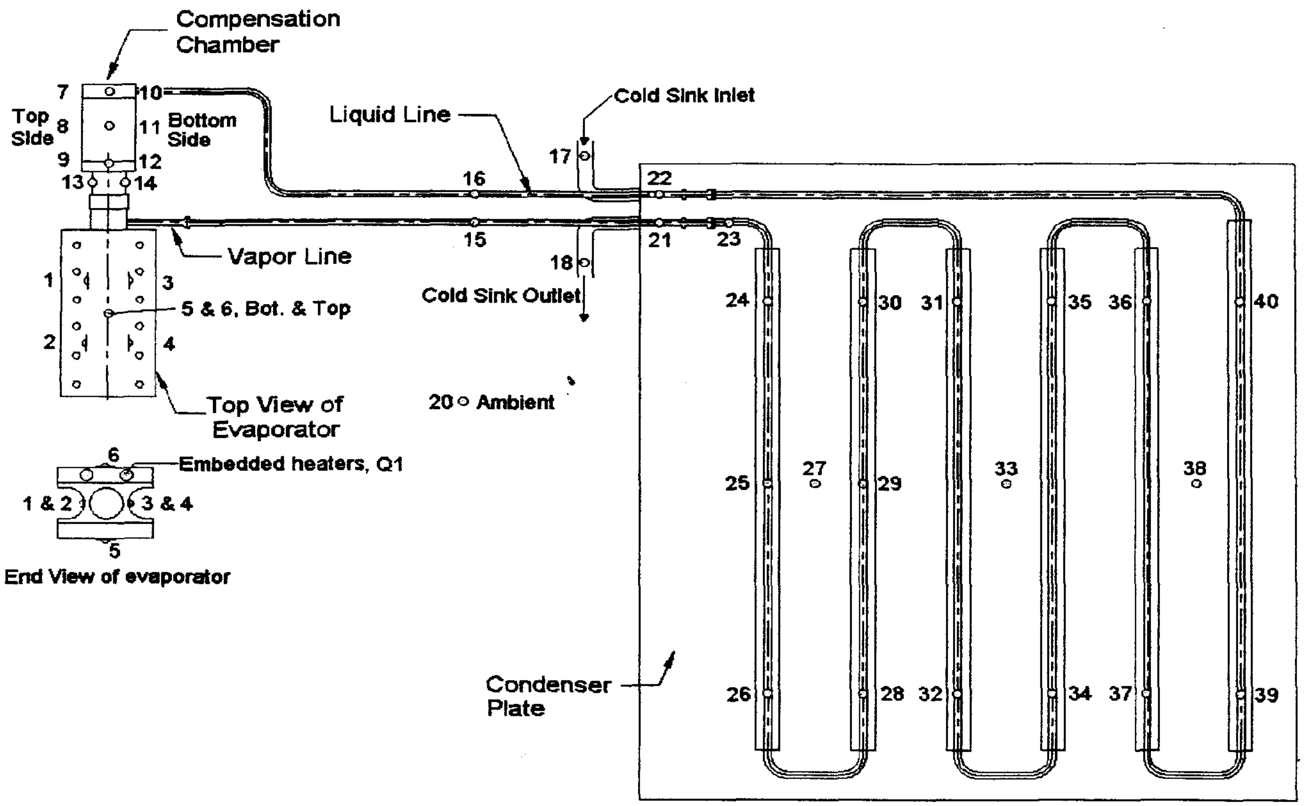

Figure 1. Schematic of Loop Heat Pipe with location of thermocouples.

condition the compensation chamber between runs so that each run started with flooded vapor grooves in the evaporator.

The condenser was attached to the evaporator assembly through a vapor line and a liquid line that were 1 m long. The serpentine condenser line was mounted on an aluminum panel. A heat exchanger identical to the condenser line was mounted on the back side of the aluminum panel. This heat exchanger was connected to a regulated thermal sink. The heat sink loop used methanol as a circulating fluid and had capacity to remove up to 200 watts at –80°C. The condenser and heat sink assembly was covered with a 5 cm thick layer of fiberglass insulation to thermally isolate it from the ambient environment. The insulated assembly was then placed inside a structural box. A dry nitrogen purge of the condenser box was used to reduce water condensation and/or freezing on the condenser panel during the experiments.

A LabView™ data acquisition program was written to perform the following operations: (1) collect the thermocouple and heater power readings at specified intervals, (2) control the power supplies operating the evaporator and compensation chamber heaters and (3) command the heat sink temperature set point. In these experiments, temperature and heater power measurements were taken every thirty seconds and recorded in a data file. The data acquisition program allowed each experimental run to step through various evaporator and compensation chamber power settings and heat sink set points. The time interval between set point changes could be changed during an experiment to compensate for variability in reaching equilibrium.

EXPERIMENTAL PLAN

The first set of experiments was performed with the ammonia filled LHP. The evaporator heat load was varied from 20 to 800 watts for condenser sink temperatures of 20, 10, 0 and –11°C. This test range would encompass both the variable conductance and the constant conductance regions of the LHP. Power was not applied to the compensation chamber heater during any of the tests reported in this paper.

The tests were conducted at fixed sink temperatures for each run while collecting data at several evaporator heat loads. The sink temperature on the condenser was allowed to reach equilibrium before heat would be applied to the evaporator. Temperature data was collected during the system start up period. When the evaporator heaters were activated, the system was allowed to reach and remain in equilibrium for a minimum of one hour before moving on to the next heater setting.

The LHP was tested in a horizontal orientation such that the evaporator and the condenser were at the same

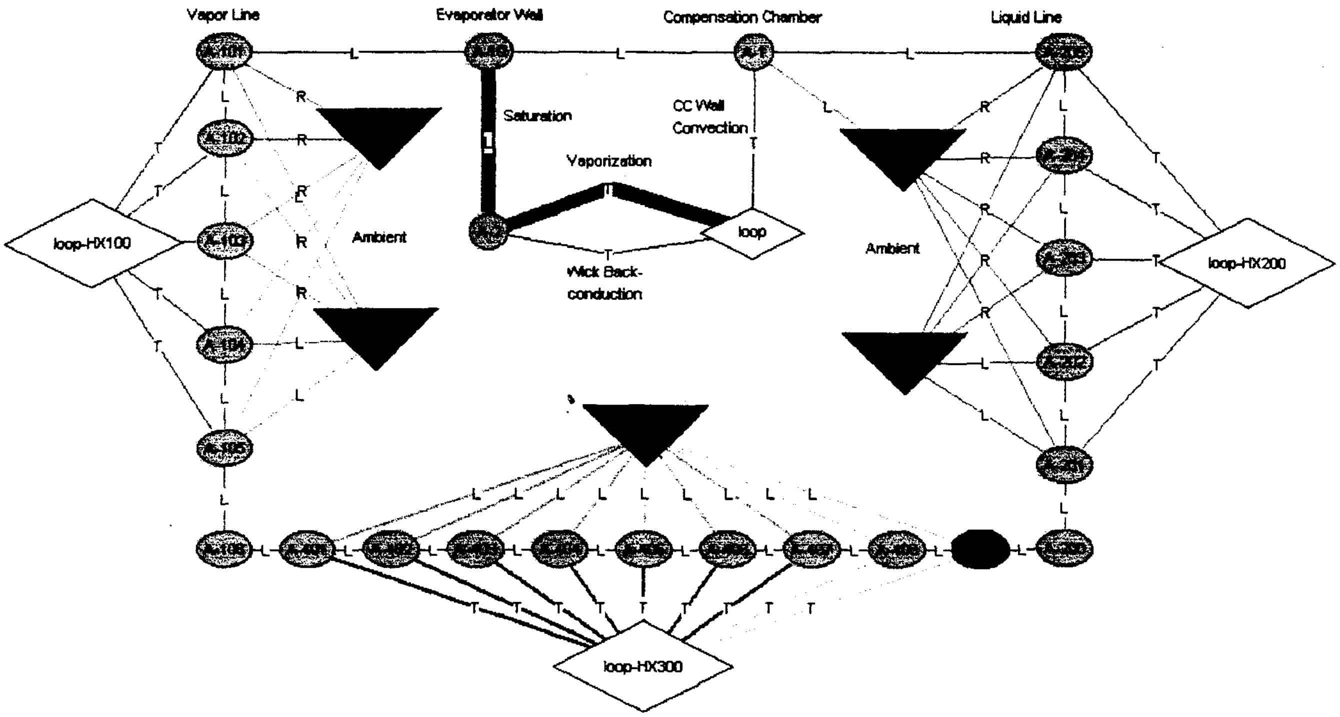

Figure 2. Schematic of Line Diagram Model for Loop Heat Pipe developed by Cullimore & Ring Technologies Inc.

elevation. This is simulates a 0-g condition since there are no gravitational body forces imposed on the capillary head to pump the working fluid through the system.

When the tests with ammonia were complete, the LHP was returned to the manufacturer to be evacuated, cleaned and recharged with propylene. The propylene tests were conducted in a similar manner as the ammonia tests except that the maximum evaporator power was limited to 200 watts. Furthermore, tests were also conducted with a vertical orientation such that the evaporator was located above the condenser. This is an adverse orientation because the capillary pressure rise must overcome the gravitational body force to circulate the propylene.

At the end of each run, the compensation chamber heater was set to five watts for about 10 minutes. It is believed that this procedure would force liquid from the compensation chamber into the vapor grooves of the evaporator wick. This was done to establish a baseline history for the next experimental run that would be identical for all runs. Ku [6] points out that there are four possible starting conditions for a LHP and that the condition with liquid flooded vapor grooves in the evaporator is the most difficult to start. It was desirable, in this test program, to determine if the test LHP would start from the most demanding condition.

NODAL NETWORK MODEL

A public domain mathematical model of loop heat pipe operation, developed by Cullimore & Ring Technology Inc., was evaluated with the data collected by this experimental program. The nodal network model uses a SINAPS*Plus*™ interface with Sinda/Fluint and is shown in Figure 2. This figure shows the thermal submodel which represents the thermal properties of each component in the LHP. The fluid submodel, which represents the state properties of the working fluid is connected to the thermal submodel as indicated in Fig. 2.

The model describes the compensation chamber with node A-1 and the evaporator with nodes A-2 and A-10. Node A-2 represents the wick OD and yields the saturation temperature. This limited number of nodes appears to be sufficient for modeling the temperature response of these elements using a wick interface. The vapor line and the liquid line are modeled with 5 nodes each and are linked to the ambient environment with both radiation and convection heat transfer modes available.

These lines were not insulated during the ammonia experiments thus a convection coefficient of 16 to 20 W/m^2K was used in the ammonia models. During the propylene experiments, the lines were insulated and a convection coefficient of 9 to 11 W/m^2K was used for the propylene models. This accounted for the additional resistance to heat flow due to the insulation on the transport lines.

The condenser is modeled with 9 nodes connected to the sink temperature. Node A-106 is the condenser inlet and A-200 is the condenser outlet.

An average conductance, G, between the condenser line and the thermal sink for the model was calculated for both working fluids from experimental data using the following expression:

$$G = Qcond/LMTDcond \qquad 1.$$

where Qcond is the heat transfer rate in the condenser and the LMTD is computed from the inlet and outlet temperatures of the condenser line and the thermal sink. The heat transfer rate in the condenser was measured in some early experiments and was found to be within 97% the heat load on the evaporator. This indicated that parasitic heat losses or gains in the system were not significant.

In the ammonia models, the value for G ranged from 29 to 33 W/K. For the propylene models the value for G ranged from 37 to 40 W/K. The sink conductance was higher for the propylene case because the condenser panel was rebuilt to reduce the thermal contact resistance between the heat exchanger lines and the substrate panel. This was accomplished by placing a thin graphfoil sheet between the lines and the condenser panel.

Table 1 describes some of the key input parameters for the LHP model based upon the geometry of the tested LHP.

DISCUSSION OF RESULTS

The performance curves of the ammonia filled LHP at four different sink temperatures are shown in Figure 3. The propylene curves are shown in Figure 4. The data presented in these figures are for the LHP tested in the horizontal (0-g) orientation. The performance curves are based on the evaporator temperature rather than the usual plot of saturation temperature because we are more concerned about the evaporator temperature on the TES instrument. In these curves, the evaporator temperature was computed from the average of thermocouples numbered 1-4 from Fig. 1. In general, these four thermocouples always indicated a temperature reading within 0.1°C of each other. The ambient temperature of the laboratory for all experiments ranged from 19 to 22°C. For each individual run the ambient temperature varied no more than ± 1°C during the course of a data collection period.

Modeling Evaporator Performance

The steady state LHP model fits the data well. For the ammonia model, the average difference between the observed and the predicted temperature was 0.85°C. The standard deviation between the model and the data was 1.2°C. This large deviation is due to the significant difference in modeling the −11°C sink data near the minimum evaporator temperature. All other data points were more accurately predicted. For the propylene model, the average difference in temperatures was 0.86°C with a standard deviation of 0.5°C. The model has the most difficulty in predicting the evaporator temperature in the variable conductance region. This difficulty is due to variations in the conductance from the sink to the condenser as the evaporator load changes. This could be improved by changing the model to accommodate the variations in the sink conductance as the evaporator load changes. As a consequence of this model short coming, the propylene data was more difficult to model than the ammonia because most of the propylene data was in the variable conductance region.

Table 1. Key Model Input Parameters

PARAMETER	VALUE
Evaporator	
length	0.152 m
diameter	0.024 m
material	aluminum
Charge mass	
ammonia	96.4 g
propylene	80.0 g
Wick	
porosity	0.60
pore size	1.2e-6 m
permeability	4.0e-14 m^2
material	sintered nickel
Compensation Chamber	
Volume	115 cm^3
material	stainless steel
Condenser	
length	3.8 m
diam., ID	4.1 mm
diam., OD	5.5 mm
material	aluminum
Transport Lines	
Length	1.04 m
ID	4.5 mm
wall	1 mm
material	stainless steel

The thermal conductance between the condenser line and the thermal sink changes in the variable conductance region because the location of the phase transition in the condenser changes. At low heat loads on the evaporator, the phase transition may occur close to the condenser inlet and allow a significant length of condenser to subcool the liquid. As the heat load on the evaporator increases, the phase change region moves toward the end of the condenser line.

LHP Conductance

In the performance curves shown in Figs. 3 and 4, the variable conductance region is limited to the curved portion of the data and the constant conductance portion is identified by the straight portion. The unit conductance of the LHP for both working fluids is shown in Figure 5 for sink temperatures of 20, 10 and 0°C.

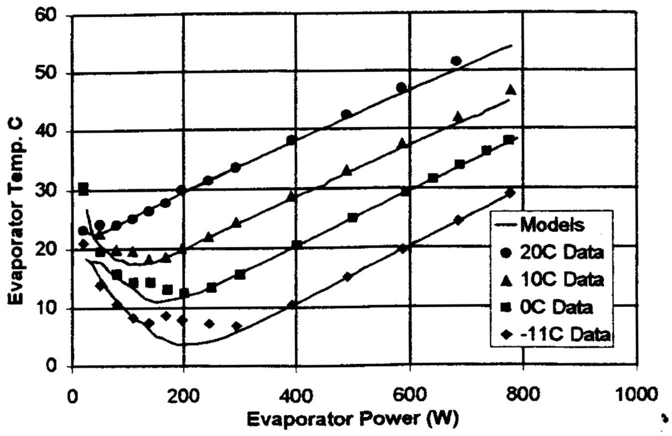

Figure 3. Performance of ammonia filled LHP superimposed with model predictions.

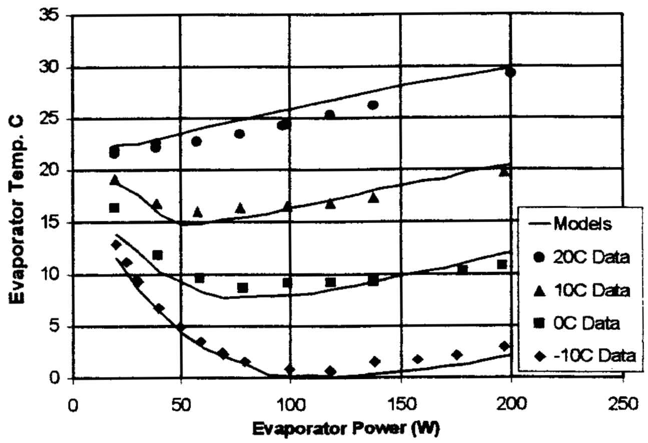

Figure 4. Performance of propylene filled LHP superimposed with model predictions.

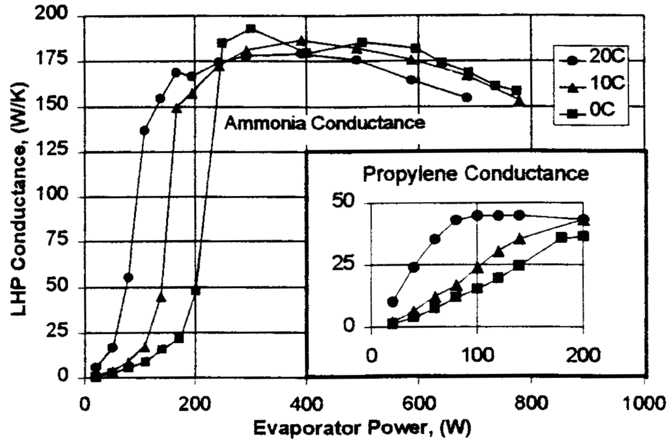

Figure 5. Comparison of conductance of ammonia and propylene Loop Heat Pipes

This figure more clearly shows where the transition from variable to constant conductance occurs.

The conductance for the ammonia LHP was 170 W/K ± 10 W/K. For the propylene LHP, the conductance was 44 ± 1 W/K. Thus the performance of the ammonia LHP is nearly four times better than with propylene as a working fluid. The ammonia LHP was tested in the constant conductance region for all four sink temperatures. For the propylene LHP, only the 20°C sink test operated in the constant conductance region. The conductance of the LHP is defined by evaporator heat load divided by the temperature difference between the condenser and the evaporator.

When the sink temperature is at the ambient temperature there is no reserve available for liquid subcooling. Thus the saturation temperature of the liquid in the evaporator must increase as the applied power increases. It is evident that the evaporator temperature varies linearly with the heat load for both working fluids. The LHP operates as a constant conductance device under these conditions.

When the sink temperature is less than the ambient temperature the evaporator performance curve possesses a local minimum that shifts toward higher heat loads as the sink temperature decreases. At low evaporator loads, the condenser can supply a significant amount of subcooling to the liquid as it returns to the compensation chamber. This in turn reduces the evaporator saturation temperature.

There are two factors contributing to the phenomenon of variable conductance in a Loop Heat Pipe: (1) the working fluid flow rate and (2) the position of the vapor/liquid phase transition in the condenser. As the phase transition moves toward the end of the condenser, the amount of liquid subcooling decreases. It is commonly observed that the evaporator temperature initially decreases as the applied heat load increases. This can be explained by noting that as the heat load initially increases, the mass flow rate of the working fluid through the system increases (up to some maximum value). This increases the overall thermal capacity of the circulating loop and causes a drop in the evaporator temperature. The observed effect is an increase in the overall thermal conductance of the LHP as shown in Figure 5.

Since there is a limit on the fluid flow rate through the system, its thermal capacity is also limited. Consequently, as the heat load on the evaporator continues to increase, the rate at which the evaporator temperature changes decreases. When the mass flow rate is maximized and saturated liquid conditions exist at the end of the condenser, the LHP enters the constant conductance region. The only way for the system to reject additional heat under these conditions is to increase the system operating temperature.

LHP Orientation

The propylene LHP was tested in both a horizontal and a vertical orientation. In the horizontal orientation the evaporator was at the same elevation as the condenser, thus there was no hydrostatic pressure on the system for the capillary pressure to overcome. In the vertical orientation, the evaporator was positioned above the condenser such that there was approximately 1 m of hydrostatic head imposed on the capillary pressure. Both orientations were tested with a sink temperature of –10°C. Under the test conditions, the LHP was in the variable conduction regime.

The effect of orientation on the performance of the evaporator is shown in Figure 6. Increasing the pressure on the capillary pump by elevating the evaporator above the condenser caused an increase in the evaporator temperature in the variable conductance region. If the LHP was operated in the constant conductance region, the evaporator temperature would not change with respect to orientation [7]. This observation is supported by the results shown in Fig. 6 where it appears that the performance curves for both orientations are converging as they approach the constant conductance region. It should be noted that the data in Figure 6 are all in the variable conductance region. From Figure 5 it can be concluded that the constant conductance region starts above evaporator loads of 200 watts when the sink temperature is at –10°C.

The observed increase in the evaporator temperature as the evaporator is elevated above the condenser is caused by the following phenomena. An increase in the pressure drop across the wick requires an increase in the temperature difference across the wick. This causes an increase in the heat conduction from the evaporator to the compensation chamber. The increased heat transport to the compensation chamber elevates the saturation temperature. The saturation temperature ultimately controls the temperature of each component in the LHP.

The data for both curves were generated by increasing the evaporator power during each test run except at power levels less than 30 watts. When the evaporator power was set to values less than 30 watts, the LHP did not start in the vertical orientation. To collect steady state data at low powers, it was necessary to start with 30 watts and step down to low power levels. One test in the vertical orientation started with a 200 watt evaporator load followed by decreasing evaporator loads. A hysteresis was observed in the power curve as shown in Figure 6 where the arrows on the vertical curve indicate data taken at increasing or decreasing power levels. The model of the evaporator power curve predicted temperatures that were between those observed in the hysteresis region.

LHP Start up

The propylene LHP had difficulty in successfully starting at low evaporator loads (below 30W) in either orientation when it was preconditioned to have liquid filling the vapor grooves. Figure 7 shows the start up behavior of

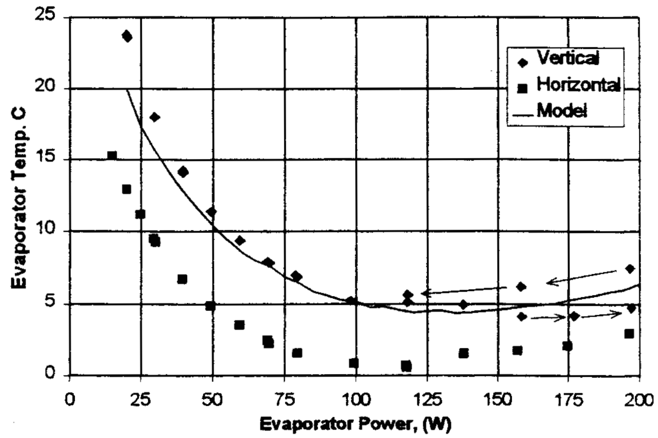

Figure 6. LHP performance comparison with horizontal and vertical orientations. Sink temperature is –10C.

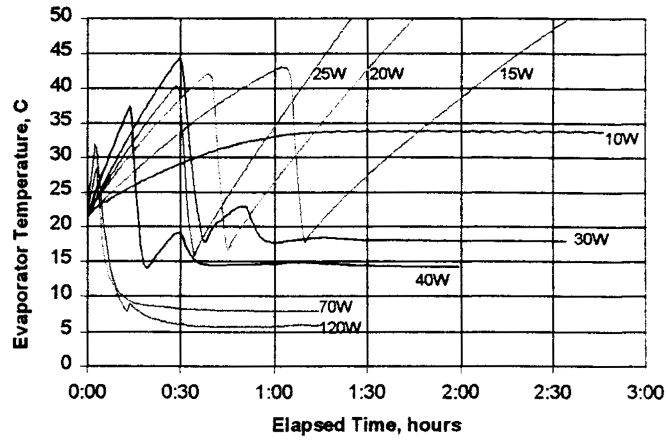

Figure 7. Evaporator response at start-up for propylene LHP in a vertical orientation with evaporator above condenser.

the LHP in the vertical orientation for heat loads ranging from 10 to 100 watts. At power levels of 15 to 25 watts, the temperature of the evaporator rose approximately 22°C before initiating what appears to be a false start. In the "false start", the evaporator temperature drops rapidly as expected in a usual starting condition but it bottoms out and the evaporator temperature experiences a second rise in temperature. The evaporator was not able to reinitiate boiling at the wick interface during the second temperature rise.

At power levels of 30 and 40 watts, the LHP successfully started but the evaporator experienced a temporary rise in temperature after the LHP started. It is interesting to note that when the evaporator temperature bottoms out after starting, this minimum temperature is about the same as the steady state evaporator temperature.

It is observed that as the heat input to the evaporator increases, the rate at which the temperature rises in the evaporator increases. The heat capacity of the evaporator body was found to be about 30 J/K. This was calculated by dividing the input power by the slope to the temperature rise observed for each condition.

CONCLUSIONS

A prototype Loop Heat Pipe has been tested with both ammonia and propylene as a working fluid with no mass attached to the evaporator. The constant thermal conductance of the ammonia filled LHP was approximately four times greater than that of the propylene filled LHP. This performance degradation was acceptable for the design of the TES instrument as a means of avoiding problems posed by cold radiators which could freeze ammonia within the condenser lines.

The LHP prebuilt model from Cullimore and Ring Technologies Inc. accurately predicted evaporator temperatures especially in the constant conductance operating region. Better modeling in the variable conductance region could be achieved by changing the sink conductance as the evaporator load changes.

Issues regarding the effect of mass on the evaporator and start-up conditions are discussed in a companion paper [1].

ACKNOWLEDGMENTS

The authors express their sincere thanks to Mr. Arthur Na-Nakornpanom for his work in setting up and operating the ammonia experiments reported in this paper. The research described in this publication was carried out by the Jet Propulsion Laboratory, California Institute of Technology, under a contract with the National Aeronautics and Space Administration.

REFERENCES

1. Rodriguez, J. I., Pauken, M. T., Na-Nakornpanom, A., "Transient Characterization of a Propylene Loop Heat Pipe During Start-up and Shut-down," SAE paper No. 2000-01-XXXX, July 2000.
2. Rodriguez, J. I., "Thermal design of the Tropospheric Emission Spectrometer Instrument," SAE paper No. 2000-01-XXXX, July 2000.
3. Maidanik, Y., F., Fershtater, Y.F., and Solodovnik, N. N., "Design and Investigation of Regulation of Loop Heat Pipes for Terrestrial and Space Applications," SAE Technical Paper Series, Paper No. 941407, June 1994.
4. Wolf A., D., and Bienert W. B., "Investigation of Temperature Control Characteristics of Loop Heat Pipes," SAE Technical Paper Series, Paper No. 941576, June 1994.
5. Nikitkin, M., and Cullimore, B., "CPL and LHP Technologies: What are the Differences, What are the Similarities?," SAE Technical Paper Series, Paper No. 981587, July 1998.
6. Ku, J., "Operating Characteristics of Loop Heat Pipes," SAE Paper No. 1999-01-2007, July 1999.
7. Kaya, T., and Ku, J., "A Parametric Study of Performance Characteristics of Loop Heat Pipes," SAE Paper No. 1999-01-2006, July 1999.
8. Wrenn, K. R., Allen, R. D., *et al.* "Verification of a Transient Loop Heat Pipe Model," SAE Paper No. 1999-01-2010, July 1999.
9. Baumann, J., Cullimore, B. et al., "Noncondensible Gas, Mass and Adverse Tilt Effects on the Start-up of Loop Heat Pipes," SAE Paper No. 1999-01-2048.

CONTACT:

For additional information contact: Dr. Mike Pauken at the Jet Propulsion Laboratory. Phone: (818) 354-4242. Email: Michael.T.Pauken@jpl.nasa.gov